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(71) Applicant *(for all designated States except US)*: IFIELD ENGINEERING PTY. LIMITED [AU/AU]; 45 Carters Road, Dural, NSW 2158 (AU).

(72) Inventor; and

(75) Inventor/Applicant *(for US only)* : FRAZER, Hugh, Ivo [AU/AU]; Cliftonville Road, Lower Portland, NSW 2756 (AU).

(74) Agent: F.B. RICE & CO.; 28a Montague Street, Balmain, NSW 2041 (AU).



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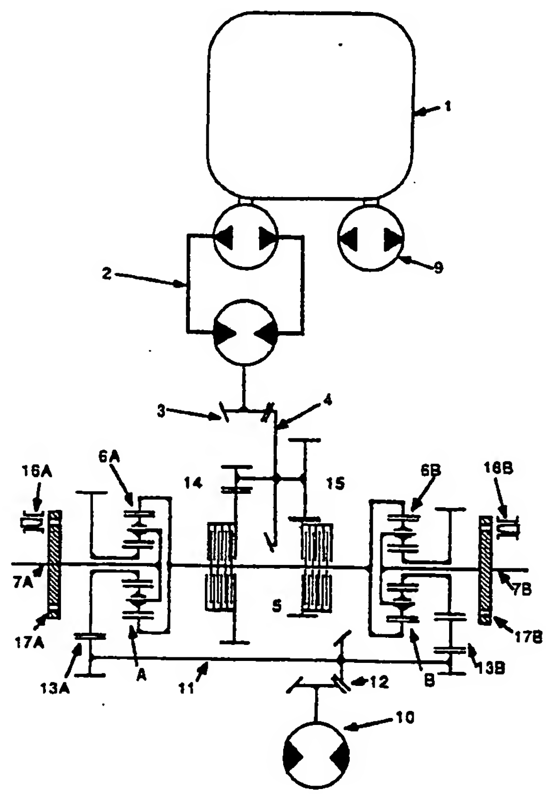
(54) Title: TRANSMISSION BRAKING SYSTEM


(57) Abstract

A transmission system for a skid-steered vehicle in which at least one hydraulic pump (9) is driven by at least one engine (1), these in turn driving a pair of hydraulic motors one connected to each of the vehicles tracks or wheels through a gear box providing at least two drive ranges each range being engageable by operation of a clutch (14, 15), the clutches (14, 15) being operable independently to select a desired range or operable simultaneously to effect emergency braking of the tracks or wheels.

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"TRANSMISSION BRAKING SYSTEM"

The invention relates to transmissions of the kind required to provide traction and steering to skid-steer vehicles, particularly militiary tracked vehicles. Such vehicles are sometimes very heavy (up to say 60 tonnes), require to have maximum agility and be capable of operating at high speeds (say 80 km/hr). The transmission then requires to have a wide ratio range, to provide for efficient regeneration of power from the inner track to the outer track during a high speed turn, to provide for accurate and precise steering, and to provide safe and reliable means for braking, all within a minimum package size and weight.

It is known that those skid-steer vehicles that use brakes on each track for steering purposes (see Paquini US Patent 3,899,058 for example) can provide braking by simultaneous actuation of the steering brakes. However such steering sytems are only suitable for low-speed crawler drives and the like, as the use of such non-regenerative steering means is too inefficient to be used on high speed skid-steer applications such as military tanks.

Modern high-speed skid-steer vehicles mostly use a separate brake on each track (for example see Tuck US Patent 3,590,658). This adds considerably to the cost and size of the transmission. Efficiency and fuel economy are also impaired as the wet clutch packs, used in order to reduce size as much as possible, have significant viscous drag when disengaged.

A further disadvantage is that the brakes fight the steering, particularly under low speed tight cornering. Under severe circumstances, such as negotiating down a steep bank while turning, the brakes can operate in opposite directions because the inner track is required to reverse to provide the turn while the outer track is

required to continue in forward. This situation reduces the effectiveness of the steering control while imposing unnecessary stress on the transmission components.

The invention applies particularly to hydrostatic transmissions consisting of one or more hydraulic pumps driven by one or more engines, that drive two hydraulic motors that in turn drive one or more gearboxes that incorporate range changing clutches. In such a system, normal braking can be provided by the hydrostatic system, but in higher ranges such braking will be adequate for normal use but may be insufficient for emergency vehicle stops. Emergency braking is also required in the event of a failure of the hydrostatic system or of its associated control system.

The object of the invention is to provide a more economic and compact construction than transmissions presently known and in use, while providing for the braking requirements with some improvements in efficiency and in performance when steering and braking at the same time. These objectives are achieved by using two or more of the range changing clutches to provide braking by simultaneous operation. A discrete brake is then no longer required with consequent savings in cost, weight, transmission size and a reduction in viscous drag losses.

In order that the nature of the invention may be better understood, preferred forms thereof are hereinafter described by way of example with reference to the accompanying drawings in which:-

Figure 1 shows schematically a conventional drive and steer system;

Figure 2 shows schematically a first form of the invention, using a hydrostatic transmission and two range clutches,

Figure 3 shows a second form of the invention using a preferred type of hydrostatic transmission and again two

range clutches,

Figure 4 shows diagrammatically a sytem for providing the necessary braking pressure to the clutches shown in Figures 2 and 3.

Figure 1 shows schematically a conventional drive and steer system as most commonly used in modern militiary tracked vehicles. There are many variations on this essential scheme, but the basic approach is almost universal. A prime mover 1 drives a transmission 2, usually an automatic gear changing transmission with a hydrokinetic torque converter. The output from the transmission drives bevel pinion 3, which drives crown wheel 4 attached to combining shaft 5. This shaft drives the annulus gears 6(A&B) of two epicyclic assemblies A and B. The output shafts 7(A&B) are driven by the planetary carriers and are each provided by brakes 8(A&B).

The prime mover also drives hydraulic pump 9 which is connected by pipework not shown to hydraulic motor 10. This hydraulic transmission provides the vehicle steering. The hydraulic motor drives the steering layshaft 11 through bevel gear set 12. The layshaft drives the sun gears of the epicylics A and B through gear trains 13(A&B). It can be seen that the gear train 13B has an additional idler gear to reverse the direction of rotation.

It can be seen that rotation of the hydraulic motor will cause opposite rotation of the sun gears. For example, if the vehicle is stationary, rotation of the hydraulic motor will cause the two output shafts to turn in opposite directions thus causing the vehicle to turn about its own vertical axis. The hydraulic pump normally used is of the variable overcentre type so that it may be controlled to provide motor rotation in either direction at any desired speed.

While the vehicle is being driven straight ahead, the

transmission 2 will provide drive through the gears to the output shafts 7(A&B) and the hydraulic motor 10 will be stationary. If the pump is subsequently operated to rotate the motor, this rotation will rotate the sun gears oppositely causing a speed difference between the output shafts. This speed difference will force the vehicle to turn. The rate of turn is easily and precisely controlled by control of the hydraulic pump.

During the turn, the inner track will have to skid, likely providing a negative torque to its respective output shaft, say 7B. This negative torque is mostly transmitted along combining shaft 5 to become an additional positive torque at the outer output shaft 7A. Because the path of this regeneration is only through the epicyclic gearing, there is little power loss due to efficiency losses. During a high speed turn, the power transmitted by this means from the inner track to the outer track may exceed the prime mover power by many times.

Braking of the vehicle is provided for by brakes 8(A&B) on the output shafts. These may be disc type brakes but are more commonly of the multi-plate oil immersed type. They are usually hydraulically operated using pressure generated by the gearbox lubrication pump, not shown, or by a special pump for clutch and brake operation, also not shown. Such brakes are typically sized to provide a maximum vehicle retardation of 0.5g.

Parking and emergency brakes are often provided for by parallel mechanical operation of the same brake packs. One potential disadvantage of this approach is that all braking is done on the same pair of brake packs, so a failure of the brake packs leads to a total loss of brakes.

A second disadvantage occurs when braking is required while steering. In an extreme case it may be necessary for the brakes to be applied during a hard slow speed turn, as when negotiating down a steep river bank.

Possibly the inner track is being rotated in reverse for steering purposes. With this braking system the inner brake 8B is fighting the retardation of the inner track and introducing unnecessarily high forces and losses within the transmission system. Even under less severe turns, the brakes add to the load of the steering system. Under some circumstances it may even be necessary to apply engine power to turn while braking.

Figure 2 shows an embodiment of the invention. The essential difference between this and Figure 1 is that the transmission has been replaced by a hydrostatic transmission 2 consisting of a pump and motor and associated circuitry and controls, not shown, with two range changing clutches 14,15 used to extend the range of the hydrostatic tranmsission so that smaller hydrostatic components may be used. These range clutches also provide for emergency braking by being operated simultaneously.

Clutch 14 engages low range, while clutch 15 engages high range. Because they are constrained to rotate at different speeds by their associated gearing, a braking effect will occur if they are both engaged simultaneously. If low range is selected, clutch 14 may remain fully engaged while 15 is progressively operated to provide progressive braking. If high range is selected, 15 may remain fully engaged while 14 is progressively operated.

Because it is complex to provide parallel mechanical operation of both clutches for parking brakes, a sprag type brake, illustrated as pins 16(A&B) engaging holes in drive plates 17(A&B) is provided for parking purposes. Any other known form of mechanical brake would perform the same function.

This scheme provides for improved safety over the conventional system in Figure 1 in that three completely seperate and different means are provided for braking

purposes. First, the normal braking through the hydrostatic system; second, the emergency braking through simultaneous operation of clutches; and lastly, the mechanical parking brake.

The hydrostatic braking is conveniently controlled by a brake pedal (not shown), the emergency braking can be readily brought in automatically by over-travel of the pedal, and a lever can operate the parking brake. The hydraulic system to the clutches is readily made completely separate from the hydrostatic drive system, so the two systems are truly independent.

Because the emergency braking effort is applied effectively to the combining shaft 5, steering through the steering motor 10 and the epicyclic gearing is entirely unaffected by braking, and steering while braking imposes no additional stresses on the drive or steering components.

It can be argued that the provision of brakes at the output is safer because there are less components to fail in the drive line between the tracks and the brake. While this argument has some validity, in practice the reliability of gearing is more easily controlled than the reliability of friction components used in brakes. The triple braking system available with a hydraulic drive transmission is an attractive proposition.

The use of a hydrostatic main drive favours a scheme using two identical hydraulic motors that both perform drive and steer functions. A schematic diagram of such a preferred embodiment is shown on Figure 3. This basic scheme reduces the overall size of the hydraulic pumps and motors to a minimum because maximum speed and maximum turn rate are never required at the same time. A system such as shown in Figure 2 has the capability of doing both at the same time even though it can never be used, so the components can not be ever fully utilised.

The prime mover 1 now drives one or two hydraulic

pumps 18. If the hydraulic motors 20 are of variable displacement, it is possible to drive and steer the vehicle with only one variable displacement pump connected to the two motors in parallel, with pipework not shown, as steering can be achieved by setting the motors to different displacements, to suit the prevailing torque demand at each gearbox input.

Two pumps can also be used, one connected to each motor, with pipework not shown. In this case, the motors can be of fixed displacement, with steering being controlled by the relative displacements of the two pumps.

The single pump option has less parts and lower weight, although steering by motor control is much more complex than steering by pump control, and is best handled by a computer based control system. In some vehicle configurations this system can allow two smaller engines to be used each driving a single pump, with improved vehicle lay-out and utility. The transmission circuitry and controls can be readily designed so that the vehicle can be driven with just one engine operative.

The two motors drive into the combining differential 21. This is shown as a bevel differential but may of spur gear type or any other known type. The casing of the differential will rotate at the mean of the two motor speeds and this represents the traction part of the drive. This casing is connected to combining shaft 5 through either high 15 or low 14 clutches with their associated gearing.

Steering is achieved by the gear trains 22(A&B) leading from the motor outputs to the sun gears of the steering epicyclics A and B. The rate of steering is thus controlled by the difference in the motor speeds.

In this case, normal braking is easily provided by the hydraulic system. During braking the motors act as pumps, and the resultant flow can either over-run the

engine by motoring the pump or can be vented over a relief valve, not shown. Emergency braking can be provided by engaging both clutches at the same time, and parking is as before achieved by the sprags 16(A&AB).

The use of clutches for braking is feasible without any need for enlargement. Full clutch pressure is calculated to give 0.7g braking in one application, which is in excess of requirements. An emergency braking capability of 0.25g is probably acceptable. An increase in the cooling flow through the clutches will be required during braking.

Figure 4 shows a simple system to provide the necessary braking pressure to both clutches at the same time. The two clutch actuating cylinders are represented by 23 and 24. The clutch pump 25 charges the accumulator 26 through non-return valve 27. Unloading valve 28 returns the pump flow to tank 29 when the accumulator is charged to the required pressure, as set by the spring adjustment on the unloading valve. The unloading valve also acts as a relief valve protecting the pump from over-pressure.

The directional control valve has three states and is shown as solenoid operated, although it could well be operated by a lever or by other known means. With both solenoids de-energised, as shown, both clutches are connected to the return port of the solenoid valve. Actuation of either solenoid will direct accumulator pressure to the respective clutch cylinder.

Variable pressure reducing valve 31 controls the back pressure at the return port of the solenoid valve and provides the braking action. This valve acts to control the pressure in gallery 32, the pressure depending on the the force of the spring 33. This spring is progressively depressed by progressive actuation of the vehicle brake pedal, beyond the travel required for hydrostatic braking,

by means not shown. The more the spring is loaded by the brake pedal, the higher the pressure in gallery 32 will be.

Such actuation of the brake pedal with the directional control in the position shown will apply the controlled pressure to both clutches to provide a controlled and progressive brake action.

If either solenoid is activated, full accumulator pressure will continue to be applied to the respective clutch cylinder, regardless whether the braking system is operated or not. Brake operation will provide controlled pressure to the unloaded clutch cylinder to provide the required progressive braking action.

High range braking is done by the low range clutch. The high clutch is used as the brake in low range. The higher vehicle speeds achievable in high range require a greater brake thermal capacity and it is convenient that the low range clutch as indicated in Figures 2 and 3 would indeed require to have a greater capacity than the high range clutch. The smaller high range clutch will be sufficient for braking at the lower speeds achievable in low range.

This description of the invention is by way of illustration only and the same principles and operation of the invention can naturally be applied to other transmission configurations. For example, the invention is not limited to transmissions with two range clutches as the directional control valves can be readily configured to provide the necessary control of clutches for both drive and braking with more than two clutches.

CLAIMS:-

1. In a skid steered vehicle a transmission system comprising at least one hydraulic pump driven by at least one engine, a pair of hydraulic motors arranged to be driven hydrostatically by said pump or pumps, each motor being connected to the vehicle tracks or wheels through a gearbox providing at least two drive ranges each range being engageable by operation of a clutch, control means to operate each clutch independently to select a desired range, and to operate at least two clutches simultaneously to effect emergency braking of the tracks or wheels.

2. The combination claimed in claim 1, including a foot operable brake pedal arranged to actuate the hydrostatic system for normal braking of the vehicle and arranged, on depression beyond a predetermined point, to actuate said control means to provide emergency braking.

3. The combination claimed in claim 2, wherein the actuation of said control means to provide emergency braking is progressive with movement of the brake pedal beyond said predetermined point.

4. The combination claimed in any one of the preceding claims wherein the two motors are connected to the tracks or wheels through a single gearbox including gearing providing for steering of the vehicle and for at least drive ranges.

5. The combination claimed in claim 4, wherein the said gearing providing for steering is arranged to bypass said gearing providing for high and low ranges.

6. The combination claimed in any one of the preceding claims, wherein said control means consists in a hydraulic cylinder arranged to actuate each range clutch, hydraulic pump means arranged to supply fluid under pressure to said hydraulic cylinders through a directional control valve, means for operating said valve to apply said fluid under pressure to either one of or none of said hydraulic

cylinders, a return line for hydraulic fluid from said valve, a variable pressure reducing valve in said return line and means to control said reducing valve whereby operation thereof causes at least partial actuation of whichever of said hydraulic cylinders are connected to said return line of said directional control valve.

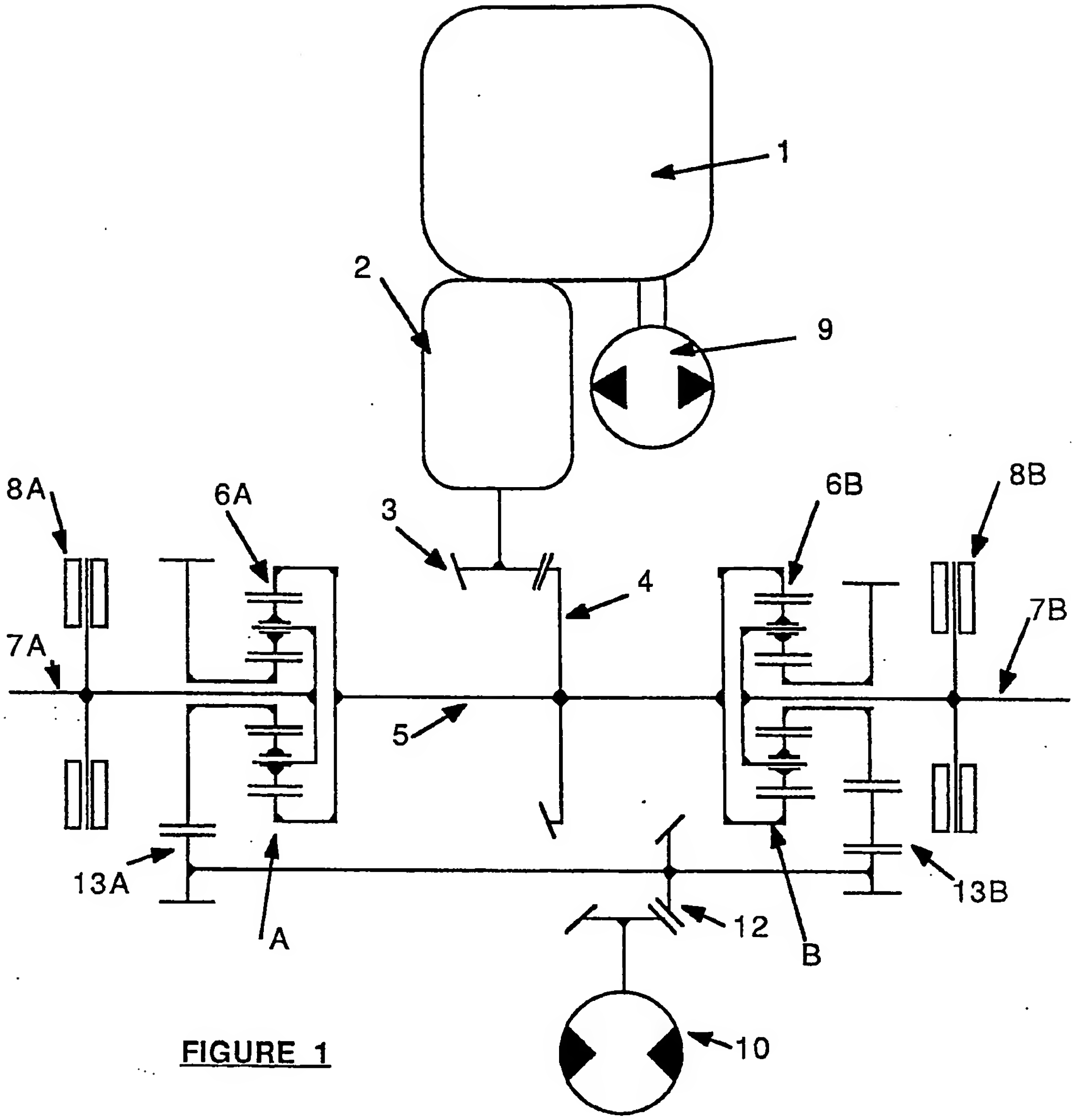


FIGURE 1

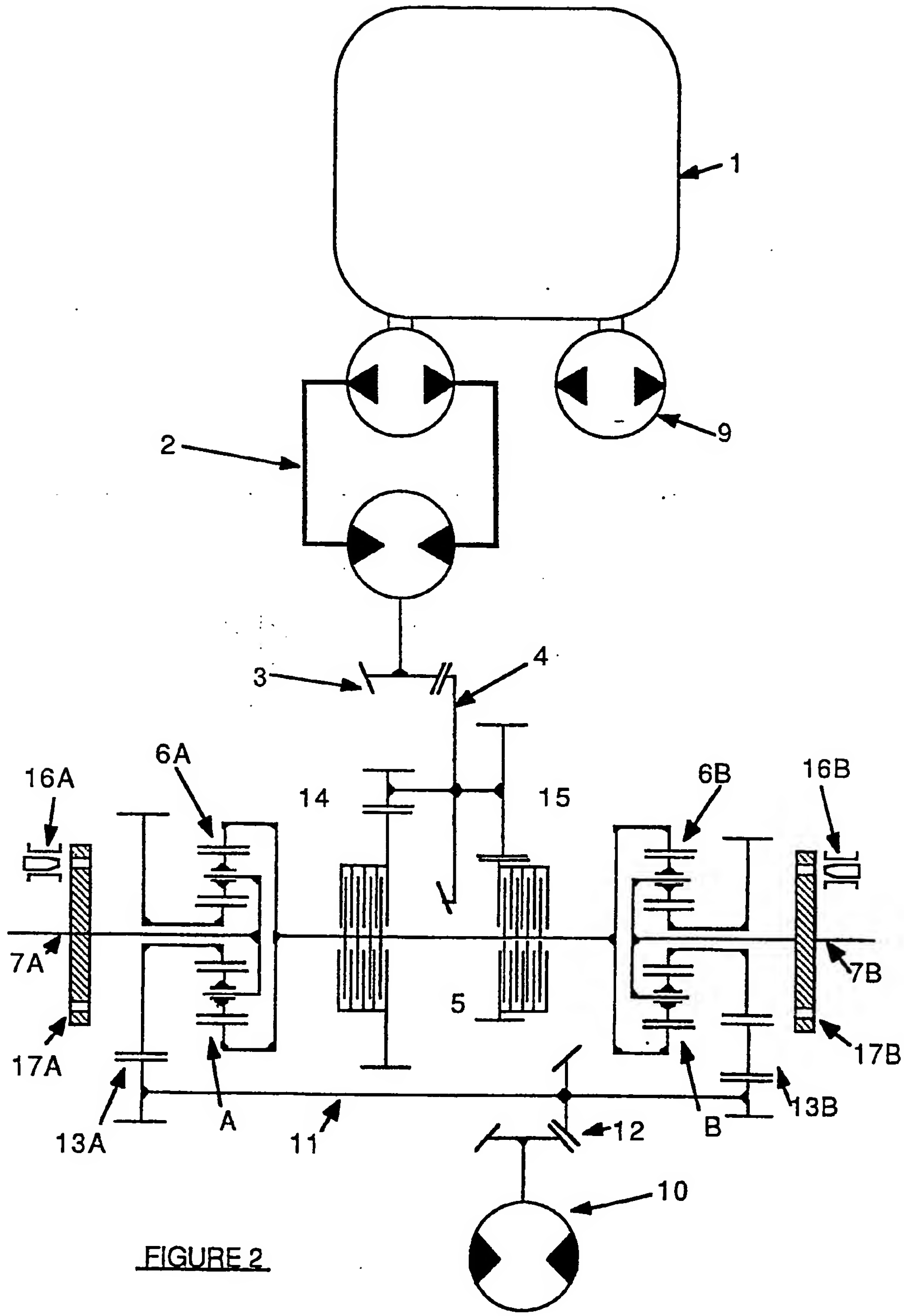


FIGURE 2

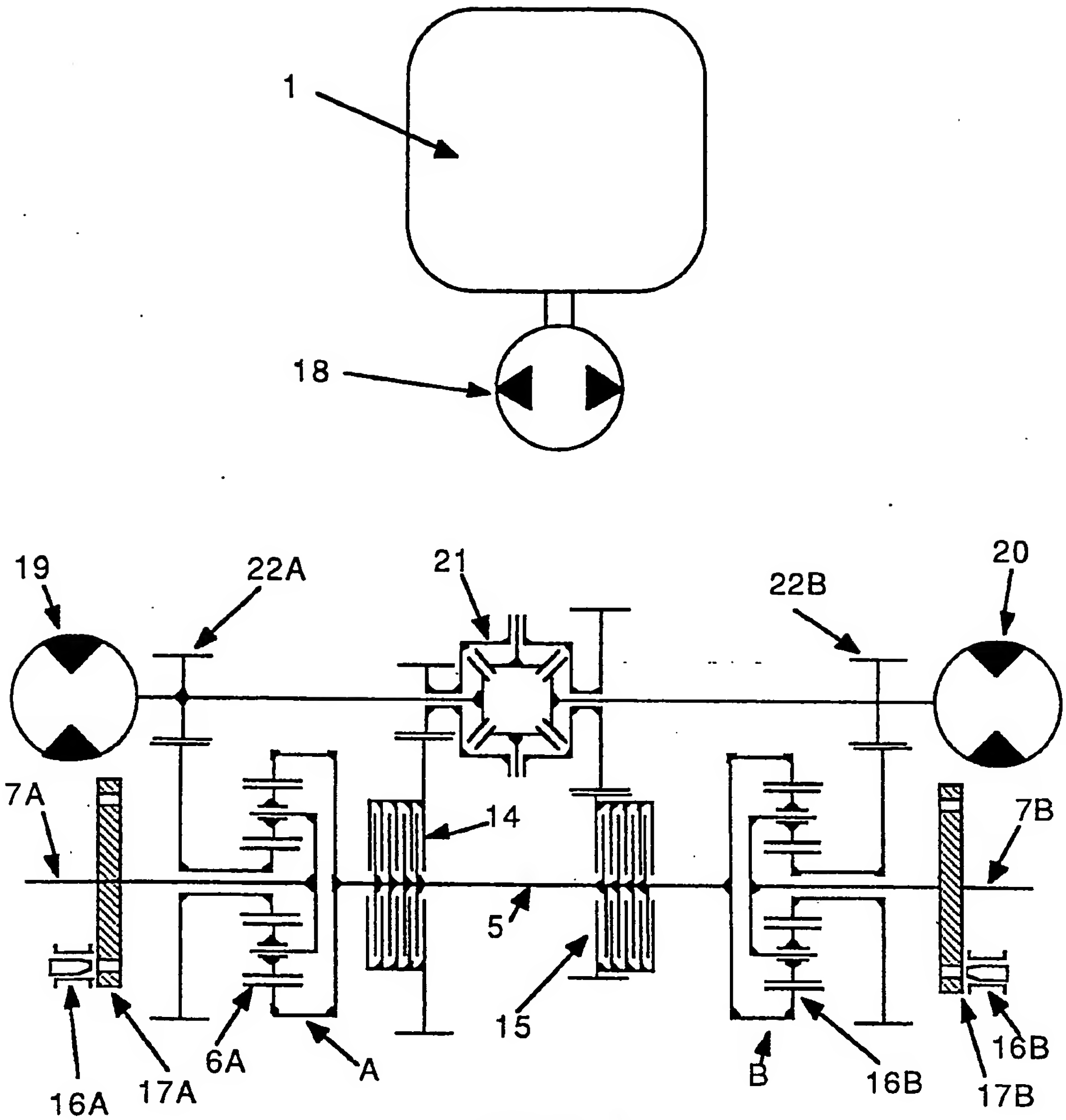


FIGURE 3

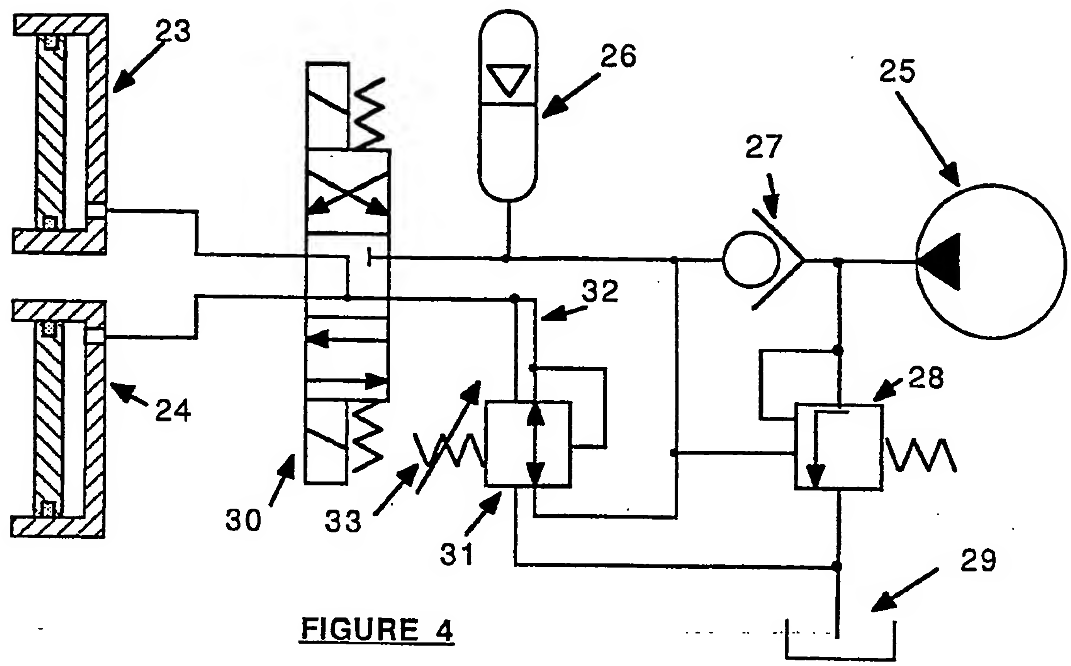


FIGURE 4

# INTERNATIONAL SEARCH REPORT

International Application No PCT/AU 87/00066

**I. CLASSIFICATION OF SUBJECT MATTER** (if several classification symbols apply, indicate all) [6]

According to International Patent Classification (IPC) or to both National Classification and IPC

Int. Cl.[4] B62D 11/12

**II. FIELDS SEARCHED**

Minimum Documentation Searched [7]

| Classification System | Classification Symbols |
|---|---|
| IPC | B62D 11/12 |

Documentation Searched other than Minimum Documentation to the Extent that such Documents are Included in the Fields Searched [8]

AU: IPC as above

**III. DOCUMENTS CONSIDERED TO BE RELEVANT** [9]

| Category [*] | Citation of Document, [11] with indication, where appropriate, of the relevant passages [12] | Relevant to Claim No. [13] |
|---|---|---|
| A | AU,B, 32651/78 (513090) (COMMERCIAL SHEARING INC.) 2 August 1979 (02.08.79) | 1-6 |
| A | AU,B, 20578/76 (512190) (FIAT-ALLIS CONSTRUCTION ~~MACHINERY~~, INC.) 22 June 1978 (22.06.78) | 1-6 |
| X | AU,B, 13372/66 (402002) (THE SECRETARY OF STATE FOR DEFENCE IN HER BRITANNIC MAJESTY'S GOVERNMENT OF THE UNITED KINGDOM) 2 May 1968 (02.05.68) | 1,4-6 |
| X | US,A, 4215755 (WATERWORTH et al) 5 August 1980 (05.08.80) | 1,4-6 |
| X | US,A, 3785450 (SUZUKI) 15 January 1974 (15.01.74) | 1-6 |

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**IV. CERTIFICATION**

| Date of the Actual Completion of the International Search | Date of Mailing of this International Search Report |
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| 25 May 1987 (25.05.87) | (09.06.87) 9 JUNE 1987 |
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ANNEX TO THE INTERNATIONAL SEARCH REPORT ON

INTERNATIONAL APPLICATION NO. PCT/AU 87/00066

This Annex lists the known "A" publication level patent family members relating to the patent documents cited in the above-mentioned international search report. The Australian Patent Office is in no way liable for these particulars which are merely given for the purpose of information.

| Patent Document Cited in Search Report | Patent Family Members | | |
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| AU 32651/78 | US 3602515<br>DE 2802979<br>US 4086767 | BR 7800426<br>FR 2377924<br>ZA 7800401 | CA 1051315<br>GB 1567105 |
| AU 20578/76 | BR 7608360<br>JP 52075734 | CA 1072610<br>US 4037677 | DE 2656625<br>US 4137944 |
| US 4215755 | GB 2006899 | | |

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